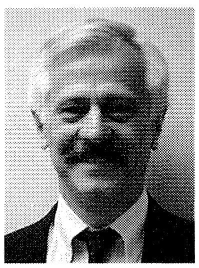
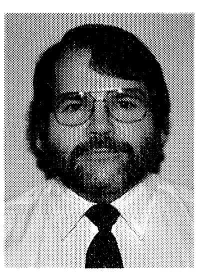
OPTIONS FOR LOW SPEED AND OPERATING SPEED BALANCING OF ROTATING EQUIPMENT

by
Carroll "Chet" Stroh
John MacKenzie
and
Jordan Rebstock
TurboCare
Houston, Texas



Carroll Stroh is Director of Engineering with TurboCare, Houston Facility, a division of Demag Delaval Turbomachinery Corporation. He has over 25 years experience in the rotating equipment business. Mr. Stroh started his career with Westinghouse Large Steam Turbine Division when it was located in Lester, Pennsylvania. While there, he was instrumental in bringing the results of their turbine research into the design process. He left Westinghouse to join DuPont and moved to their Beaumont, Texas plant where he consulted on turbomachinery problems in plants throughout the Gulf coast. After five years in the field, he moved to Wilmington, Delaware to the DuPont Experimental Station where he developed his expertise in rotordynamics. He spent the rest of his career with DuPont acting as a Consultant's Consultant and provided computer simulation of equipment to aid in the troubleshooting process.

Mr. Stroh has authored and coauthored several papers on rotordynamic behavior. He earned a B.S. degree, an M.S. degree, and did three years of post graduate work in Mechanical Engineering at the University of Pennsylvania. He also has a B.A. degree (Mathematics) from Lebanon Valley College. He is a member of Tau Beta Pi.



John R. MacKenzie received his B.S. degree (Engineering Technology) from the University of Houston (1982). His major was in Mechanical Design. He has 21 years experience at the TurboCare Houston Facility. The last 18 years Mr. McKenzie has headed up the Operating Speed Balance Facility in Houston. During this time, he has been involved with the balancing of 2000 plus rotors from over 31 different manufacturers.

Jordan Rebstock received a B.S. degree (Electro-Optics) from the University of Houston-Clear Lake (1988). He initially started with vibration instrumentation repair, installation, and calibration, which led to rotating equipment repair in the Gulf Coast area. He has worked in the industry for eight years as a field and shop balancer, vibration technician, balance engineer, and project manager. He has been with TurboCare for four years as a balance engineer and project manager.

ABSTRACT

Most turbomachinery users refer to bearing vibration as one of their major indicators of a successful repair. Frequently heard are such comments as "we could not even tell the machine was running." The single most observed cause for vibration is rotor unbalance, yet sometimes this is not supported with the proper level of importance placed on balance work. It sometimes seems that more importance is placed on the exterior paint job. This tutorial focuses on options available for balancing turbomachinery. The strong and weak points of the various options available are pointed out and several case histories are highlighted.

Though balance is important, it should not be considered a cure for all problems. If major problems exist, they need to be taken care of before any balance work is done.

BALANCING LANGUAGE

Unbalance is usually measured by the product of the radius times weight such as gram-in or rotor eccentricity. To get a feel for how it works when making balance corrections, either a large weight can be added at a small radius or a proportionately smaller weight at a larger radius. To talk about imbalance in terms of weight only without radius is meaningless. The term rotor eccentricity refers to how far the total rotor weight would have to be offset from the axis of rotation to produce the indicated unbalance amount. For instance to find the gram-in required, multiply the total rotor weight in grams by the given eccentricity in inches. The angular location of the correction weight will be discussed later when some general balancing approaches are discussed.

Vibration amplitudes refer to vibration occurring only at the running speed frequency which enables us to focus only on rotor imbalance. The vibration is measured using a stationary pickup so as the rotor whirls from the unbalance, the stationary pickup sees vibration occur as a sine wave. The maximum amplitude is used for the vibration measurement and the phase angle refers to the lag between when peak amplitude is reached and a mark on the shaft passes a reference point.

RIGID ROTOR

A rigid rotor might be defined as any rotor which runs at least 25 percent below its first flexible rotor critical speed so there is no significant rotor deflection. There are cases where a rotor might run above what is called a rigid rotor bearing critical speed, all the motion takes place in the bearing clearance with no rotor deflection. This falls under the description of a rigid rotor. A rigid rotor can be balanced by means of correction in any two separate planes regardless of how the unbalance is distributed throughout the rotor. For such cases, low speed balancing can satisfy all balancing requirements.

FLEXIBLE ROTOR

A flexible rotor might be defined as a rotor which runs above a flexible rotor critical speed or within 25 percent below that speed with sufficient rotor deflection to alter rotor balance behavior. Once a rotor starts to deflect, the actual axial distribution of imbalance becomes important and corrections have to be made in more than two planes, close to where the actual unbalance is located. Low speed data alone cannot supply all the necessary information on an assembled rotor to accomplish a complete balance of a flexible rotor. If the rotor can be stripped down to the bare shaft, a progressive stack and balance can be done, more about this will be discussed later. But, if the rotor is an integral forging of discs and shaft this is not possible. There are ways to mathematically come up with a best guess for making corrections at low speed, but above certain levels there is no guarantee the results will be acceptable.

Occasionally the queston is asked why certain integral rotors which run above a critical speed and have never been operating speed balanced still run with little vibration. There are several possible explanations for this. If an integral rotor has only two major components such as two discs, using these two discs as the correction planes will likely place the corrections where the most probable actual unbalance is located. For more complex rotor configurations, a possible explanation is that users know where their rotor critical speeds are and ramp through them very quickly so there is no time for the system to respond to the critical. This practice is fine as long as there are not seals that need to be broken in during the startup. Another possible explanation is that some rotor bearing systems differ in their degree of sensitivity to unbalance. Even though the rotor deflects there is still sufficient motion in the bearings to provide adequate system damping. For such relatively insensitive systems, a best guess balancing approach has proven quite satisfactory over the years, but not all rotor bearings systems are this forgiving and the key is to know the difference.

ROTOR LIFE OUTSIDE THE MACHINE

Storage. Rotors stored horizontally are prone to taking a temporary set from gravity sag. Usually this can be relieved by running the rotor in the balance stand until the level of unbalance reaches a steady state level. Do not start balancing until steady state is achieved. Storing rotors vertically is preferred.

Shipping. Here, too, due to bouncing over bumps, a stackable rotor could take a set, this may or may not come out when run in a low speed stand. The only way to know if something has changed is by comparing with prior runout and balance records.

Cleaning. Grit blasting a rotor is known to affect rotor balance. Any cleaning of the rotor in this manner should be done before balancing.

Preservatives. Cleaning the rotor of preservatives completely is difficult. For rotors sensitive to unbalance, storing them in canisters with a nitrogen purge should be considered.

GRAVITY SAG

In contrast to a set bow the rotor may take after storage in a horizontal position, there is a natural gravity sag in the rotor. When the rotor turns, it turns about this natural sag curve, the rotor does not straighten as it is sometimes argued. This means that focusing on a spot on the rotor, the surface goes through alternating tensile and compressive stresses as the rotor rotates. Contrary to intuition, the natural gravity sag of the rotor has no effect on the rotor balance.

SHRUNK ON PARTS

On most "built up" rotors, wheels, discs, impellers, sleeves are mounted on the shaft with a predetermined interference fit. The minimum is dictated by a requirement for metal to metal contact when at operating speed. Should a clearance exist at speed, the part goes offcenter and an unbalance is created. Typically parts are designed to have much higher amounts of interference than the minimum, but not always. On parts with keys the interference might be light.

One experience was with an eleven stage pump rotor. The shaft was quite flexible and the part interference was just enough to hold the shaft in a deflected position if pushed off center. The rotor had been shipped to a shop for a low speed check and was found to have a large static unbalance. Of course any time a large static is found the next step should be a shaft TIR check. The TIR results agreed with the balance stand findings so further exploration was done and the rotor was impacted on the high spot. A recheck showed a 180 degree shift in the location of the high spot. At this point, a light pressure was applied to the high spot and the shaft straightened out. No balance corrections were made. This points out that it is very important to know the condition and behavior of a rotor before doing any balance work. Any correction made while the rotor was so deflected can become a built in unbalance if the rotor straightens when at operating speed. Long slender pump shafts rely on the stiffening effects of the impeller wear rings, they are not intended to be run to speed in air.

DRIVE COUPLING UNBALANCE

Any time a drive coupling with mass is attached to the shaft, there is the potential for driver unbalance to alter rotor unbalance. The only way to assure that this is not the case is to do a coupling reversal before ending the balance work. This means physically disconnecting the driver coupling, rotating it 180 degrees and reattaching it. If there is no driver unbalance, the results should be the same as the previous run. If there is a change, make half the correction on the driver and half on the rotor to achieve the correct distribution. A belt drive is much preferred for low speed balancing because it eliminates the possibility of this problem.

DRIVE KEYS

Keys should be made to fill the shaft and the drive coupling hub, nothing more. A good check is to record the half key weight used to balance the shaft and the half key weight used to balance the hub and be sure the sum of these two values equals the weight of the key used. Most rotors are quite sensitive to coupling unbalance and on more than one occasion, an incorrect key weight caused a problem.

PROGRESSIVE STACK AND BALANCE

When a stackable rotor is involved, a progressive stack and balance is possible. By this is meant that starting with a bare shaft, parts can be mounted two at a time with a balance correction being made after each pair is installed. In this way the balance correction is made directly on the part which has introduced an additional unbalance. All this works quite well but achieving successful results depends heavily on not introducing any shaft deflections which might be relieved when the unit is put in operation. It should be pointed out that the amount of deflection allowed is quite small. The balance stand is much more sensitive to shaft deflection than a dial indicator. This is not to say that this approach does not work, it does work and has been used many times, but shaft runout must be given a lot of attention. What is overlooked could be a big surprise.

MANDREL BALANCING WHEELS/IMPELLERS

When balancing wheels for overspeed testing, a mandrel is used to support the wheel during the balancing process. Too large a mandrel can change the balance characteristics of the wheel along with consume all of the balance tolerance sought. When balancing, the mandrel should be pre-balanced before the wheel is installed. During wheel installation, a lot of care is needed to assure that the wheel sits on the mandrel correctly. It is not unusual for a shrunk on part to grab the shaft and not sit squarely. Balancing a wheel in this state will result in a built in unbalance.

Should mandrel balancing each wheel beforehand be done and how important is a couple in the wheel? Frequently, wheels are mandrel balanced with most of the time spent removing any couple unbalance. The usual small plane separation on wheels makes this difficult. After mandrel balancing, the wheels are mounted on the shaft and for any unbalance after assembly, a single plane correction is made to the wheel. This cancels the work done to eliminate the couple in the mandrel balance. So how important is it to remove the couple in the first place? When the differences between a rigid rotor and a flexible rotor are discussed, it all has to do with how readily the rotor deflects from the unbalance. For a reasonably thick shaft and close wheel spacing, dynamic couples in wheels has little significance. For an overhung construction with some distance between the overhung wheel and the bearing, dynamic couples are very important. If the rotor to be balanced is somewhere between configuration extremes, than maybe a closer look is necessary to evaluate the importance of couples.

OVERHUNG ROTORS

Balancing impellers or wheels of an overhung design by placing the wheels on a mandrel and balancing between bearings is not equivalent to balancing in the overhung state. Why this is the case is unclear. A rotor is more sensitive to unbalance from an overhung weight and possibly this translates into greater sensitivity in the balancing machine. Modern balancing machines can handle wheels in an overhung position by simply using the proper machine set up.

BALANCING TOLERANCES

A frequently used balance tolerance is API 617 2.9.5.2

$$\text{Umax} = 4\text{w/n per plane} \tag{1}$$

The above equation for unbalance tolerance in ounce-in is four times rotor weight in pounds divided by the rotor in service operating speed in rpm. If a two plane correction is to be made, as is usually the case, the tolerance should be divided by two for the tolerance in each plane.

RESIDUAL UNBALANCE CHECK

Residual unbalance as per API 617 Appendix D is the amount of unbalance remaining in a rotor after the balancing. This is a time consuming process that validates the balancing machine calibration and confirms the balancing technician has done the job properly. The procedure involves placing a known weight that is between one to two times the unbalance tolerance used during balancing in each correction plane used. The known weight is then moved and recorded at six to 12 different angular locations, all at the same radius. The residual unbalance is then plotted with the balancing machine response on the y-axis and the angular location of the weight on the x-axis. The residual unbalance test is to be done on every correction plane used. If the final residual unbalance in any of these planes is greater than the original tolerance, then a more precise balance is necessary.

It is worth noting that implied in the equations for computing residual unbalance is the fact that the trial weight be larger than the residual unbalance left in the rotor. This means that when the weight is placed 180 degrees from the heavy spot the phase angle will change by 180 degrees. If by some error the residual unbalance left in the rotor is more than two times the tolerance sought, the API equations will produce erroneous results.

SOFT BEARING AND HARD BEARING BALANCE MACHINES

In low speed balance work, there is the choice between these two machine types. The soft bearing machine is very sensitive to unbalance and it might be argued that a higher level of balance can be achieved with light rotors. The hard bearing machine has the distinct advantage in that it can be calibrated to read directly in the amount of unbalance and correct for cross plane interaction. A disadvantage of hard bearing machines, which is more dependent on the user than the machine, is in cases of small plane separation. Typically, a couple in two planes separated by one inch or less will look as bad on the screen as planes with much greater separation. The force on the bearings is much more affected by wide plane separation. A lot of time and money is wasted on correcting couples on close plane separation which have little impact on rotor behavior.

MODAL BALANCING

Modal balancing is simply placing balance corrections so they only affect the mode of interest. For instance, with a symmetrical rotor to excite only the first U mode place 10 ounces at zero degrees at the exact rotor midspan and five ounces at zero degrees on either side of the mid span halfway between the mid span and the bearing location. Such an arrangement will not excite the second S shape mode. To excite the S mode and not the first U mode, simply take away the rotor midspan correction and arrange the five ounces on either side of the mid span so they are 180 degrees out-of-phase. In a similar matter, other higher modes can also be addressed. In this way correction for the first U mode can be made and then move on to the second mode without disturbing the first mode.

INFLUENCE COEFFICIENT BALANCING

Low Speed Balancing

First take vibration data at each of the two bearings in terms of synchronous vibration levels (filtered for only running speed vibration) and phase angle. Then by using a known trial weight and placing it at known location, first in the one balance plane and then in the other, sufficient information is obtained to determine the corrections required in each of these two planes to balance the rotor. This is done by simultaneously solving the resulting vectorial equations. The process is referred to as the influence coefficient method because by using known weights at known locations, their influence can be measured and a two by two array of the resulting influence coefficients can be set up which, when multiplied with the original vibration data, yields the corrections required.

Operating Speed Balancing

A similar process is used when the rotor runs above a critical speed. The only difference is that now we need to obtain an array of influence coefficients at more than one speed and more than two planes of correction are now required.

OPERATING SPEED BALANCING SOFTWARE

The second author started in the operating speed balance field some 18 years ago. He spent the early part of his career in trying to locate all available articles on the theory of balancing rotors at operating speed. Over the span of about a year he collected over 65 articles. The most notable was a paper written on a least squares influence coefficient theory. The paper focuses on applying a least squares approach to minimize the resulting residual unbalance. This paper was used to write a simple balancing program and the program is used today when needed.

After nearly three years of balancing rotors, the authors are not completely happy with any of the available balancing software they have tried. If the nature of the balancing work involves doing the same style rotor over and over again, as in an OEM environment, the available software might be adapted to the need. But when every balance job is different, such as in a repair facility, the present software to satisfy all needs has not been found. Most rotors are balanced by plotting on polar paper the resulting vectors of each trial weight. The authors feel they have more control of what is going on during the balancing process. And as their experience has grown, they have gotten bold enough to start grinding the permanent corrections on the rotor before making the corrections with temporary balance weights.

The authors balance a rotor by plotting vectors on polar paper. They find the final results are usually better than with the computer programs they have tried. Over the span of some 18 years of balancing rotors at operating speed, and having balanced over 2300 rotors, they feel the results of this approach speak for themselves. Again because of the diverse range of rotor styles they deal with, they prefer the influence coefficient approach to balancing flexible rotors over modal theory. In the modal theory of balancing rotors either N or N plus two correction planes are used. N is the number of criticals the rotor goes through before reaching operating speed. Modal theory requires making simultaneous corrections in multiple planes based on the mode shapes of the rotor being balanced. This usually requires more planes than would be used with the influence coefficient approach. They have balanced rotors that operate above the first critical and close to the second critical using only two planes and in a few occasions, only a single plane was necessary. They have yet to get a balancing program to balance a rotor with an out of tolerance static unbalance at it's operating speed. They find this to be the hardest type of unbalance to correct in a flexible rotor because of the influence of such a correction on the rotor's first critical speed. If the correction lowers the operating speed response, it usually increases the response at the first critical speed and vice versa.

The authors are in the process of purchasing the latest balancing program technology to go along with an instrumentation upgrade. They are hopeful this new software will meet their more versatile needs.

FIELD BALANCING

If the owner chooses not to balance in a shop setting and wishes to do so in the field, numerous problems can arise:

- Frequent stopping and starting of the driver. The driver is sometimes involved with the process requiring at least a partial startup of the plant. Electric motors are limited to the number of starts in a certain time interval.
- Balancing personnel safety is a concern due to the variables of the environment.
- Correction weight placement or removal might be difficult. They may not always have proper access for making actual correction.
- Have to tolerate influence of other factors causing vibration such as misalignment, natural frequencies of support structure or other equipment in the area, air churning during the balancing process, feedback from the rest of the flow path when not in the operating state, difficulties around whether the unit is loaded and to what degree, sometimes high temperature and/or chemicals prevent access, and need room for vibration and phase measuring equipment.
- Balancing process takes longer and sometimes there is not adequate time for the job.
- Field balancing is frequently attempted to correct other mechanical deficiencies.
- Not immediately obvious that the problem is unbalance. Several days can be lost while trying to be sure balancing is the answer.

EXAMPLES OF LOW SPEED BALANCE

Six Stage Compressor Rotor Balance

A six stage compressor was pulled from service to be cleaned and balanced. After cleaning, the rotor was to be disassembled and progressively restacked and balanced. Two wheels were to be assembled at a time and each wheel just installed was used as one of the balancing planes for a two plane low speed correction. The wheels were not mandrel balanced and no effort was made to evaluate any couple unbalance in any of the wheels. This was a case of close wheel spacing and a rather thick shaft as discussed earlier under MANDREL BALANCING WHEELS/IMPELLERS.

Constant monitoring for shaft deflection is a very important part of this process because it indicates if a wheel is installed in a cocked position on the shaft. Any additional shaft run-out caused by a cocked wheel along with any couple added by the cocked wheel contributes more unbalance. If these unbalance adders in the cocked state are corrected and the wheel straightens at operating speed, there is a built-in unbalance. So the whole process can be somewhat tedious and time consuming but many users have successfully balanced stacked rotors in this way.

Some additional observations in this case might be of interest. The plant that owned the above rotor always installed the spare during shutdowns and sent the rotor just removed from service into the shop for cleaning and balance. So after the work was done the rotor was prepared for storage by spraying with a preservative and shipped roughly 100 miles by truck to the plant. One year later, when the rotor was to be used, the plant would always send the rotor back to the shop for solvent cleaning and a check balance. Unbalance readings were never the same as recorded when the rotor left the shop the year before. The rotor was usually within tolerance but the readings were different. Part of the answer was probably due to some preservative that remained after cleaning but the truck ride was suspected of causing parts to move or settle.

Turbine Balance

An eleven stage turbine that runs at 4200 rpm and has a first critical at 3200 rpm was successfully low speed balanced using an approximate correction distribution technique. The approach is based on representing the unbalance distribution in a rotor as a sum of the rotor modal contributions. The final results for a between bearing rotor are approximately 25 percent-50 percent-25 percent distributed correction for a static unbalance and using the end planes for the full couple correction with a rotor that runs above the first critical. It can be shown by considering a worse case scenario, that the final balance will be equal to, or less than, a predictable percentage of the initial unbalance and if that level is acceptable than low speed balancing can proceed. If not, then an operating speed balance or a progressive stack and balance, if appropriate, will be necessary.

EXAMPLES OF OPERATING SPEED BALANCE

Frame 1 Type Gas Turbine

The machine is a gas turbine with a 2000 lb rotor consisting of an axial compressor body supported between bearings and a single turbine wheel overhung on one end. The rotor was balanced at a speed of 11,140 rpm. The bearing span is 79 in and the journals are 4.125 in in diameter. Running in the bunker pedestals, the rotor has a first critical speed between 6600 and 7100 rpm with the rotor ends nearly in-phase and a split second critical at 8800 rpm and 11000 rpm with the rotor ends out-of-phase. The initial run of this rotor produced the highest vibration of any rotor ever run in our balance bunker. Prior to the operating speed balance work, the plant contact had interacted with the OEM for over a year and a half trying to make the rotor run acceptably. All field attempts to resolve the problems failed. The engineer who recommended the purchase of the unit was given field responsibility to make the rotor run. The rotor was finally shipped to Houston for balancing. It took a total of 76 runs and three days to get the rotor to the best balance condition achieveable. The normal 1.0 mm/sec balance tolerance could not be reached. Final results of the balance were 3.4 and 3.9 mm/sec at 11,140 rpm, the operating speed. The rotor was installed in the field and surprisingly ran under 1.0 mil shaft vibration in the case, indicating a looser balance tolerance may be appropriate for rotors of this design.

Five Stage Compressor Rotor

The machine is a five stage compressor. The rotor was balanced at a speed of 4150 rpm. The rotor weighs about 7000 pounds with a bearing span of about 110 in. The journals are 5.0 in in diameter. The rotor runs in the bunker pedestals with a first critical speed at 3600 rpm. The rotor was balanced with a single center plane correction at the third stage. The author gave a talk at the Vibration Institute monthly meeting in Houston in the early eighties and referenced this exact data. Upon discussing this data, it was immediately pointed out by persons in attendance, that according to theory, it takes three planes to balance a rotor that operates above it's first critical speed. In general this would be the case, but the results speak for themselves. Obviously if all the unbalance happens to be in the third stage, it would be foolish to insist on making additional corrections elsewhere. Webster's definition of "theory" is an assumption based on limited information. When a rotor is balanced at speed, there are no assumptions.

Seven Stage Second Body Ammonia Compressor Rotor

The machine is a seven stage second body ammonia compressor. The rotor weighs about 1500 pounds, has a bearing span of 84 in, and 3.5 in diameter journals. The rotor runs in the bunker pedestals with a first critical speed at 2200 rpm and a second critical speed at 7200 rpm. The rotor was balanced at a speed of 8900 rpm. Three planes were used to balance the rotor, both end planes and a midspan plane. It took a total of 31 runs and a day and a half to balance the rotor to normal tolerance. The rotor runs above the second critical in the balance bunker. This is a rotor that does not run very well in the bunker after assembly balance at low speeds. The results are much improved if a progressive low speed balance is done, where low speed balance corrections are made after every two wheels are installed, but operating speed balance results are by far the best.

Single Stage Overhung Compressor Rotor

The machine is a single stage, overhung compressor, see Figure 1. The rotor was balanced at its operating speed of 11,500 rpm. The rotor weighs 800 pounds and has a bearing span of about 28 in. The journals are 3.5 in in diameter. The rotor runs in the bunker pedestals with a first critical at 9600 rpm and a second critical just above 11,500 rpm. Because the compressor is overhung, the first critical has the rotor ends out-of-phase and the second critical has the ends nearly in-phase. Corrections were made permanent to the overhung compressor wheel by grinding. It took a total of eight runs to balance the rotor to the balance tolerance at operating speed. There was only one plane for correction which limited what could be done with the first critical during the balancing process.

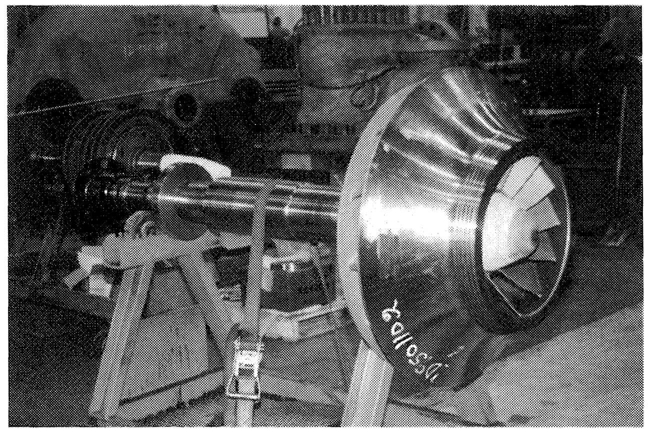


Figure 1. Overhung Compressor Rotor.

Frame 3 Type Gas Turbine

The machine is a frame 3 type gas turbine. The rotor consists of an axial compressor body between bearings and a single turbine wheel overhung on one end, see Figure 2. The rotor weighs about 4500 pounds, has a bearing span of 78 in, and journals are 6.0 in in diameter. The rotor runs in the bunker pedestals with an in-phase first critical speed at 3300 rpm and an out-of-phase second critical at 6600 rpm.

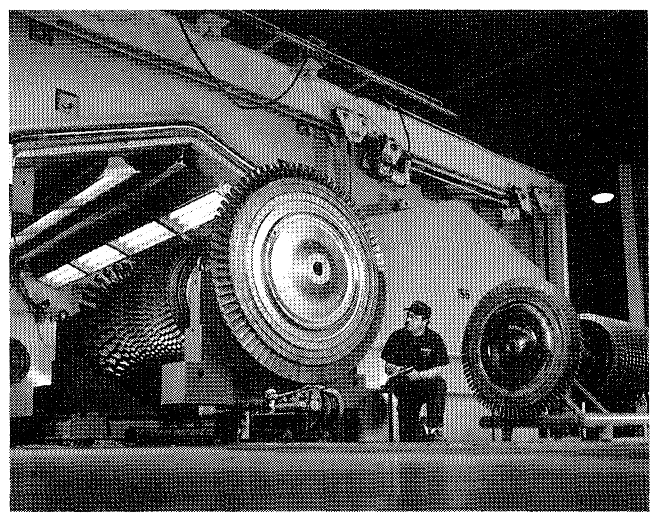


Figure 2. Frame 3 and Frame 5 Gas Turbine Rotors.

The compressor body of this rotor was first progressively stacked and balanced in a low speed facility. The rotor was then run in the bunker with no blades in the turbine wheel and balanced at its operating speed of 7500 rpm by making corrections only to the turbine wheel. The turbine wheel was then bladed and according to OEM specification would require no further balancing. The OEM professes that if a balanced set of blades are installed in a balanced rotor, then no further corrections to the rotor will be needed. This has not been found to be the case and usually further corrections are made to the turbine wheel. The results for over 15 rotors show that corrections had to be made every time.

Axial Compressor Rotor

The machine is an axial compressor, see Figure 3. The rotor was balanced at 5985 rpm using only two correction planes. The rotor weighs about 4500 pounds and the journals are 6.125 in in diameter. The rotor runs in the bunker pedestals with a first critical speed at 2400 rpm and a second critical speed at 5400+ rpm. The correction planes are located at the ends of the axial body. There were no provisions for a mid plane correction. Using the available two end planes, the unbalance to a static condition at operating speed was able to be reduced. The resulting balance was over 2.0 mm/s and not acceptable.

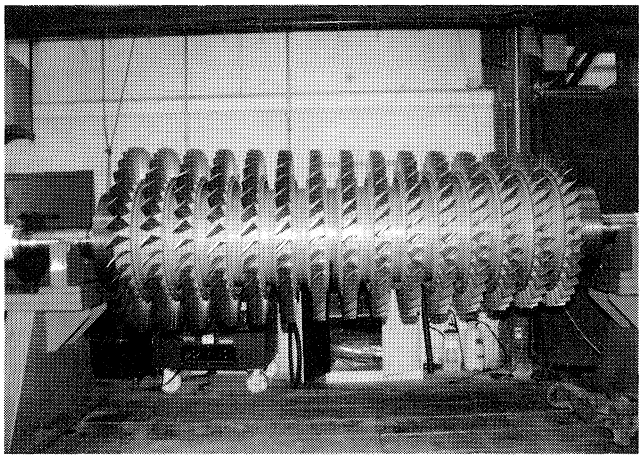


Figure 3. Axial Compressor Rotor.

The rotor was returned to the OEM repair shop and modified so a mid plane correction could be made to the rotor. The rotor was then returned to the balance bunker. The unbalance condition was finally reduced to an acceptable level by using all three correction planes. The end user accepted the balance of the rotor and the rotor was put into storage for later use. This is a good example of how proof testing a rotor at speed can reveal a needed rotor design change, in this case an additional balance plane. Had the rotor been put in service without the third plane addition, the need would have surfaced later when possibly the plant/OEM interaction may not have been realized so quickly.

Frame 5 Type Gas Turbine

The machine is a frame 5 type gas turbine, with all correction planes between the bearings, see Figure 2. The rotor weighs about 20,000 pounds and the journals are 8.0 in in diameter. Balance correction weights are added to individual circumferential slots spaced axially down the rotor. The rotor runs in the bunker pedestals with an in-phase first critical speed at 2400 rpm and a second out-of-phase critical speed at 4500 rpm. The rotor was balanced at its operating speed of 5200 rpm. Two correction planes were used, a mid plane slot and one of the end plane slots. A total of 11 balance runs were made. The time needed to make each run with a rotor this size is about 2.5 hr. Typically most rotors take 10 to 15 minutes for a run. The total time for the rotor balance in the bunker was about 30 hr, which includes the setup and take down time.

Seven Stage Compressor Rotor

The machine is a seven stage compressor. The rotor weighs 1100 pounds, has a bearing span of 72 in, and journal diameters of 3.0 in. The rotor runs in the bunker pedestals with a first critical speed at 4800 rpm. The rotor was taken to its maximum continuous speed of 10,900 rpm. Rotor unbalance was within tolerance. A repeat run was made and a run with hydraulic pedestal stiffeners was also made. There were no corrections made to the rotor. The balance condition at maximum continuous speed stayed within the standard balance tolerance. This example was given to show that corrections are not always necessary, but the user still benefits from knowing the rotor will run at speed without difficulty.

BIBLIOGRAPHY

Bishop, R. E. D. and Parkinson, A. G., "On the Use of Balancing Machines for Flexible Rotors," Journal of Engineering for Industry (May 1972).

Darlow, M. S., "Balancing of High Speed Machinery," Springer-Verlag (1989).

Den Hartog, J. P., "The Balancing of Flexible Rotors," Air, Space, and Instruments.

Drechsler, J., "A Combination of Modal Balancing and the Influence Coefficient Method," Newcastle Vibration Conference (1975).

Giers, A., "Practice of Flexible Rotor Balancing," IMECHE (1976).

Goodman, T. P., "A Least Squares Method for Computing Balance Corrections," Journal of Engineering for Industry, *86* (1964).

Gunter, E. J., Barrett, L. E., and Allaire, P. E., "Balancing of Multimass Flexible Rotors," *Proceedings of the Fifth Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1976).

Hundal, M. S. and Harker, R. J., "Balancing of Flexible Rotors Having Arbitrary Mass and Stiffness Distribution," Journal of Engineering for Industry (May 1966).

Iwatsybo, T., "The Balancing of Flexible Rotors," Bulletin of the Japanese Society of Mechanical Engineers (December 1980).

Kellenberger, W., "Should a Flexible Rotor be Balanced in 'N' or 'N+2' Planes?," Journal of Engineering for Industry (May 1975).

Lund, J. W. and Tonnesen, J., "Analysis and Experiments on Multi-Plane Balancing of a Flexible Rotor," Journal of Engineering for Industry, Paper No. 71-VIBR-94.

Parkinson, A. G., Jackson, K. L., and Bishop, R. E. D., "Experiments on the Balancing of Small Flexible Rotors: Part I–Theory," Journal Mechanical Engineering Science, *5,* (1) (1963).

Parkinson, A. G., Jackson, K. L., and Bishop, R. E. D., "Experiments on the Balancing of Small Flexible Rotors: Part II–Experiments," Journal Mechanical Engineering Science, *5,* (1) (1963).

Parkinson, A. G., Smalley, A. J., and Darlow, M. S., "Demonstration of a Unified Approach to the Balancing of Flexible Rotor," Journal of Engineering Power (1980).

Tessarzik, J. M., Badgley, R. H., and Anderson, W. J., "Flexible Rotor Balancing by the Exact Point–Speed Influence Coefficient Method," Journal of Engineering for Industry, Paper No. 71-VIBR-91.

Tessarzik, J. M., Badgley, R. H., and Fleming, D. P., "Experimental Evaluation of Multiplane–Multispeed Rotor Balancing Through Multiple Critical Speeds," Journal of Engineering for Industry (August 1976).

Tonnesen, J., "Further Experiments on Balancing of a High Speed Flexible Rotor," Journal of Engineering for Industry, Paper No. 73-DET-99.